An ultrasonic sound speed sensor for measuring EGR levels

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Abstract: Exhaust gas recirculation (EGR) has been used for years to improve the performance of internal combustion engines. This paper shows that acoustic methods can be used to measure EGR. Theory is presented which shows that measurements of the speed of sound can be used to measure the amount of EGR in the intake manifold. In particular, a new method called the discrete acoustic wave and phase detection (DAWPD) method can be used to measure EGR levels with a fast-response time. Experimental results show that a DAWPD sensor can be used to measure EGR levels with adequate accuracy ($\pm$1.3 per cent EGR) at steady state. Transient measurements were not possible owing to engine limitations. The sensor's performance was limited by the ultrasonic transducers used. It is postulated that sensor performance could be improved with smaller and temperature-independent non-resonant transducers.

Keywords: exhaust gas recirculation, exhaust gas recirculation measurement, speed of sound

1 JUSTIFICATION FOR AN EXHAUST GAS RECIRCULATION SENSOR

Exhaust gas recirculation (EGR) is a technique used to lower oxides of nitrogen (NO_x) emissions in automobiles. The principle of the technique is to mix exhaust gas with the intake air–fuel mixture. This has the effect of lowering the combustion temperature and slowing the combustion reaction, which has been shown to reduce NO_x emissions. EGR levels of 15 per cent (by mass) have been shown to reduce NO_x emissions dramatically and modern engines have been able to tolerate EGR concentrations up to 30 per cent [**1**, **2**]. To optimize fuel economy, emissions, and driveability, EGR must be controlled accurately. Without variable valve timing, as engine speed and load increases, the amount of residual gas (or 'internal EGR') left in the cylinder decreases. Too little EGR in the cylinder provides fewer emission reduction benefits, while too much EGR can result in misfires and slow burning velocities which result in a significant increase in unburnt hydrocarbons. Owing to the variations in the amount of internal EGR, an external EGR system is used to control the amount of EGR.

Generally, external EGR is controlled by connecting the exhaust manifold and the intake manifold with a pipe to which a control valve is fitted. In an open-loop control system, the valve position is set for the desired EGR level based on engine parameters, such as throttle position and engine speed. In a closed-loop control system, the valve is controlled with a valve position sensor and a pressure sensor. The differential pressure is measured across the valve so that the flowrate through the pipe can be measured. The control valve can be adjusted for the desired amount of EGR. Open-loop control of EGR can become extremely inaccurate. Open-loop control is not robust to parameter variation as changes in the open-loop system are directly seen in the output. Over time, particulate matter from the exhaust gas can clog the EGR pipe and control valve. As the valve and pipe become clogged, the amount of EGR reaching the intake manifold will be lower than expected. Closed-loop control can compensate for changes in valve performance due to clogging.

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However, the pressure measurement by the pressure sensor is only valid during steady state operation. EGR level predictions can be complicated during transient engine operation. Even for short-duration EGR changes, there can be a significant increase in unburnt hydrocarbons [**1**]. A fast-response real-time EGR level measurement method is desirable to optimize engine emissions.

There are a few measurement methods that have been used to measure EGR in real time. Real-time non-dispersive infrared (NDIR) CO_2 sensors have been used to measure the EGR levels in intake manifolds [**3**, **4**], where the volume fraction of EGR in the intake manifold is determined by

$$y_{\mathrm{EGR}} \approx \frac{y_{\mathrm{CO_{2,INTAKE}}}}{y_{\mathrm{CO_{2,EXHAUST}}}} \tag{1}$$

where $y_{\mathrm{CO_{2,INTAKE}}}$ and $y_{\mathrm{CO_{2,EXHAUST}}}$ are the volume fractions of carbon dioxide in the intake manifold and exhaust manifold respectively. (Typical concentrations of CO_2 in the atmosphere are approximately 0.04 per cent (by volume), while concentrations of CO_2 in the exhaust are typically much higher [about 12.5 per cent (by volume) for the stoichiometric combustion of octane, for example]. Therefore, this is a relatively accurate approximation.) Real-time NDIR-based methods have fast response times (of the order of 1 ms) and good accuracy [**3**]. However, NDIR sensors are extremely expensive and not feasible for production vehicles.

Another EGR measurement method is the use of a thermal anemometer, which operates on a similar principle as a hot-wire anemometer [**5**]. This sensor has shown that it has adequate accuracy and time response, but there are questions about manufacturability, cost, and long-term accuracy.

Previously, it has been shown by the present authors that the discrete acoustic wave and phase detection (DAWPD) method can be used to measure the speed of sound a gas mixture [**6**]. The DAWPD method can also be used to measure the EGR level in the intake manifold. A sensor using the DAWPD method can be inexpensive (of the order of a few US dollars in mass production) and have a very fast response time.

2 THEORY OF EGR MEASUREMENT BY SOUND SPEED

EGR in the intake manifold of an internal combustion engine can be found by measuring the speed of sound of the intake mixture. A change in the sound speed is caused by a large increase in the temperature of the intake mixture due to an increase in EGR. The relationship between EGR level and temperature can be derived by applying the conservation of mass and conservation of energy across an adiabatic mixing section (Fig. 1). For the thermodynamic system shown in Fig. 1 the conservation of mass across the mixing section is

$$\dot{m}_{\mathrm{EGR}} + \dot{m}_{\mathrm{air}} = \dot{m}_{\mathrm{mix}} \tag{2}$$

where $\dot{m}_{\mathrm{EGR}}$, $\dot{m}_{\mathrm{air}}$, and $\dot{m}_{\mathrm{mix}}$ are the mass flowrates of the EGR, air, and the mixture respectively.

Furthermore, the conservation of energy applied across the adiabatic mixing section, neglecting heat loss and potential and kinetic energy changes, results in

$$\dot{m}_{\mathrm{EGR}}h_{\mathrm{EGR}} + \dot{m}_{\mathrm{air}}h_{\mathrm{air}} = \dot{m}_{\mathrm{mix}}h_{\mathrm{mix}} \tag{3}$$

where h is the specific enthalpy.

Substituting equation (2) into equation (3) and writing in terms of the mass ratio x_{EGR} of EGR to air flow reveals

$$x_{\mathrm{EGR}} = \frac{\dot{m}_{\mathrm{EGR}}}{\dot{m}_{\mathrm{air}}} = \frac{h_{\mathrm{mix}} - h_{\mathrm{air}}}{h_{\mathrm{EGR}} - h_{\mathrm{mix}}} \tag{4}$$

For an ideal gas the change in enthalpy can be approximated by the product of the specific heat C_p at constant pressure and the change in the temperature. Furthermore, the specific heats at constant pressure of air and exhaust gas are approximately the

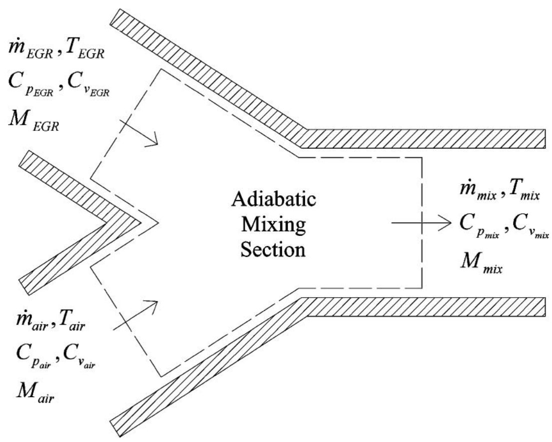

Fig. 1 Schematic diagram of the mixing of exhaust gas and fresh intake air, where $\dot{m}$ is the mass flowrate, T is the temperature, M is the molecular weight, and C_p and C_v are the specific heats at constant pressure and constant volume respectively. The subscripts EGR, air, and mix refer to the properties of the exhaust gas recirculation, the intake air, and the mixture of the EGR and air respectively

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same, allowing equation (4) to be expressed only by the temperature of the gases. Therefore, equation (4) can be simplified to

$$x_{\mathrm{EGR}} = \frac{C_{p,\mathrm{air}}(T_{\mathrm{mix}} - T_{\mathrm{air}})}{C_{p,\mathrm{EGR}}(T_{\mathrm{EGR}} - T_{\mathrm{mix}})} \approx \frac{T_{\mathrm{mix}} - T_{\mathrm{air}}}{T_{\mathrm{EGR}} - T_{\mathrm{mix}}} \tag{5}$$

Traditionally, EGR levels are expressed as the ratio of EGR mass flow to the total mass flow, or the mass fraction w_{EGR} of EGR. The relationship between x_{EGR} and w_{EGR}, which has been simplified into terms of the temperature of the gases, is given by

$$w_{\mathrm{EGR}} = \frac{\dot{m}_{\mathrm{EGR}}}{\dot{m}_{\mathrm{EGR}} + \dot{m}_{\mathrm{air}}} = \frac{x_{\mathrm{EGR}}}{1 + x_{\mathrm{EGR}}} \approx \frac{T_{\mathrm{mix}} - T_{\mathrm{air}}}{T_{\mathrm{EGR}} - T_{\mathrm{air}}} \tag{6}$$

Equation (6) is expressed graphically in Fig. 2 where the temperature of EGR and air are 500 °C and 20 °C respectively, and the exhaust gas is the product of stoichiometric combustion of methane and air. The figure shows that even small amounts of EGR can greatly increase the temperature of the intake mixture.

Since the addition of EGR to the intake manifold increases the temperature of the intake mixture, the speed of sound of the intake mixture can be used to determine the amount of EGR in the mixture. The advantage of a sound speed measurement compared with a traditional temperature measurement (i.e. thermocouples) is that sound speed measurements have much faster response times (microseconds) and are more durable.

For an ideal gas, the sound speed can be calculated from

$$c = \sqrt{\frac{\gamma R T}{M}} \tag{7}$$

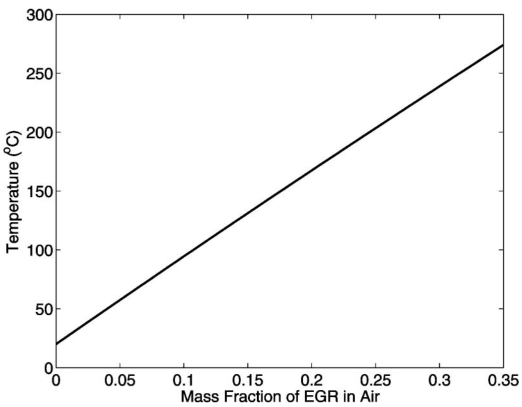

Fig. 2 Variation in intake mixture temperature with concentration of EGR at 500 °C in air at 20 °C where the exhaust gas is the product of a stoichiometric combustion of methane and air

where R is the universal gas constant, T is the temperature, M is the molecular weight, and γ is the ratio of the specific heat C_p at constant pressure to the specific heat C_v at constant volume.

For a binary mixture of intake air and EGR the sound speed can be found from

$$c = \sqrt{\frac{R T_{\mathrm{mix}}[y_{\mathrm{EGR}} C_{p,\mathrm{EGR}} + (1 - y_{\mathrm{EGR}}) C_{p,\mathrm{air}}]}{[y_{\mathrm{EGR}} M_{\mathrm{EGR}} + (1 - y_{\mathrm{EGR}}) M_{\mathrm{air}}] \times [y_{\mathrm{EGR}} C_{v,\mathrm{EGR}} + (1 - y_{\mathrm{EGR}}) C_{v,\mathrm{air}}]}} \tag{8}$$

where y_{EGR} is the volume fraction of EGR in the intake manifold.

Changes in the air–fuel ratio in engines have very little effect on the molecular mass and specific heats of the exhaust gas. Conventional spark-ignition engines will maintain a constant air–fuel ratio during engine operation; so M_{EGR}, $C_{p,\mathrm{EGR}}$, and $C_{v,\mathrm{EGR}}$ will not change. In compression ignition engines the air–fuel ratio will change with engine load. However, during extreme changes in the air–fuel ratio, the changes in M_{EGR} and γ_{EGR} are less than 1 per cent. (In the idealized stoichiometric (equivalence ratio of 1, i.e. $\Phi = 1$) combustion of cetane, the properties of the exhaust gas are $\gamma_{\Phi=1} = C_{p,\Phi=1}/C_{v,\Phi=1} = 1.369$ and $M_{\mathrm{EGR},\Phi=1} = 28.70$. In an extreme lean case ($\Phi = 0.5$), the properties of the exhaust gas are $\gamma_{\Phi=0.5} = C_{p,\Phi=0.5}/C_{v,\Phi=0.5} = 1.382$ and $M_{\mathrm{EGR},\Phi=0.5} = 28.76$. Therefore, the difference in γ/M [the ratio of interest, see equation (7)] between the stoichiometric and lean cases is 0.7 per cent.) Furthermore, equation (8) can be further simplified by the fact that the molecular mass and specific heats for EGR and air are approximately the same. With this simplification, the speed of sound of the intake mixture in a stoichiometric engine can be expressed as

$$c \approx \sqrt{\frac{C_{p,\mathrm{air}}}{C_{v,\mathrm{air}}} \frac{R T_{\mathrm{mix}}}{M_{\mathrm{air}}}} \tag{9}$$

The sound speed of a mixture of EGR and air can be expressed as a function of the fraction of EGR in the mixture when equation (6) is substituted for the mixture temperature according to

$$c \approx \sqrt{\frac{C_{p,\mathrm{air}}}{C_{v,\mathrm{air}}} \frac{R}{M_{\mathrm{air}}} [T_{\mathrm{air}} + w_{\mathrm{EGR}}(T_{\mathrm{EGR}} - T_{\mathrm{air}})]} \tag{10}$$

Figure 3 graphically represents the full model described in equation (8) and the approximation described in equation (10) for an air–EGR mixture

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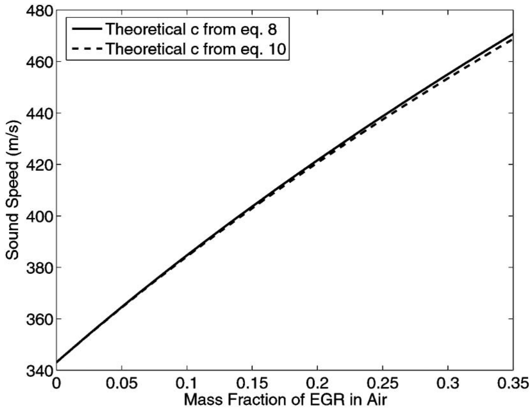

Fig. 3 Theoretical sound speed variation with concentration of EGR at 500 °C in air at 20 °C where the exhaust gas is the product of a stoichiometric combustion of methane and air. The solid curve represents the full model described in equation (8) (where the volume fraction of EGR has been converted to the mass fraction) and the dashed curve represents the approximation described in equation (10)

from the combustion of methane. The maximum difference between the two models is 1.8 per cent at 35 per cent EGR (in terms of EGR levels the difference is 0.6 per cent EGR). Therefore, the approximation in equation (10) is an accurate representation of the system. The figure also shows that even small amounts of EGR can greatly increase the speed of sound of the intake mixture, just as the temperature was shown to increase. Therefore, sound speed measurement can be used to measure external EGR in an internal combustion engine. The range of sound speeds that could be encountered in the intake manifold will depend on the temperature of the intake air, the temperature of the exhaust gas, and the EGR level. In general, the intake air can range in temperature from −40 to 40 °C, while exhaust temperatures can be as high as 700 °C. EGR levels can range from 0 to 35 per cent. An acoustic EGR sensor must be able to measure a range of sound speeds from 300 m/s (pure air at −40 °C) to 480 m/s (a mixture of air at 40 °C and 35 per cent EGR at 700 °C) as found from equation (10).

3 EXPERIMENTAL SET-UP

The theory of EGR measurement using acoustic methods was tested using the DAWPD sensor recently developed by the present authors. Previously, the DAWPD sensor was used to measure the quality of gaseous fuels in automotive applications [**6**]. The DAWPD sensor has been described in detail by Olfert [**7**]. The DAWPD method measures the phase difference between a transmitted ultrasonic have and a received ultrasonic wave, where the phase difference between the waves is a function of the sound speed. Figure 4 shows a schematic diagram of the system. The frequency generator (custom design [**7**]) produces a continuous 40 kHz square wave (which matches the resonant frequency of the piezoelectric transducers) that drives the piezoelectric transmitter (Panasonic EFR–TQB40KS). The piezoelectric transmitter converts the electrical signal into an ultrasonic vibration that travels through the gas medium and is received by the piezoelectric receiver (Panasonic EFR–RQB40K5). Likewise, the receiver converts the ultrasonic signal into an electric signal that is amplified and conditioned into a square wave (also at 40 kHz). The square wave produced by the frequency generator and the square wave produced by the receiver are fed into a phase discriminator which produces a voltage that is a function of the phase difference between the sent and received waves, where the phase difference is a function of the sound speed of the medium.

The experimental set-up is shown in Fig. 5. A Cooperative Fuel Research (CFR) engine (model CFR-48) was used for the experiment and is described in Table 1. The CFR engine was controlled with a Digalog 1022A dynamometer controller from

Table 1 CFR engine description

Specification	Value
Displacement	0.6 l
Number of cylinders	1
Compression ratio	8.68
Number of valves	2
Fuel	Natural gas

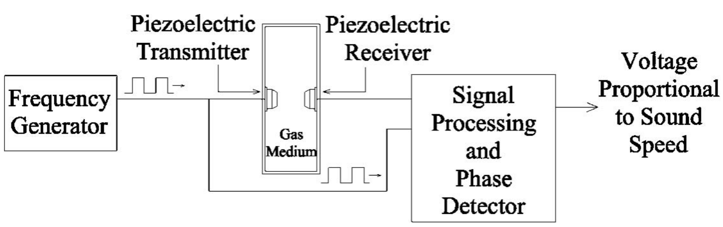

Fig. 4 Schematic diagram of the DAWPD ultrasonic gas sensor

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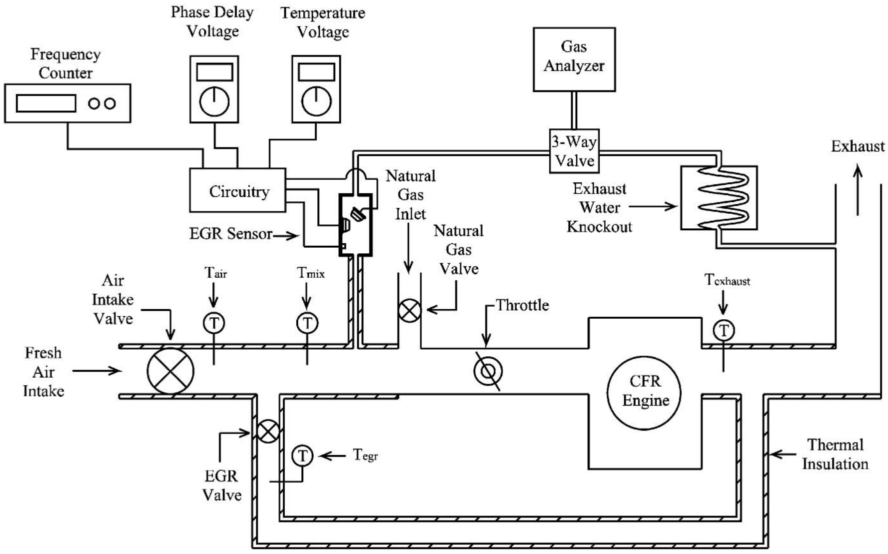

Fig. 5 Schematic diagram of the experimental set-up for measuring EGR levels with a DAWPD ultrasonic gas sensor

which the engine speed could be set for any throttle position. For this experiment the engine was run at 800 r/min. The CFR engine operated on natural gas supplied directly from the building supply. The amount of natural gas was controlled manually with a gate valve (see Fig. 5). A Snap-On MT3500 emission gas analyser measured the composition of the exhaust gas. The CFR engine ran at a stoichiometric air–fuel ratio by adjusting the fuel flowrate so that the amounts of CO and O_2 in the exhaust were minimized. For accurate CO_2 measurements with the Snap-On NDIR gas analyser, water vapour must be removed from the sample line. The infrared absorption of water vapour is similar to those of CO and CO_2; so the water vapour present in the sample will result in an inaccurate measurement with an NDIR detector. For this reason an exhaust water knockout was used for the exhaust sample. A water knockout was not needed for the intake mixture since the water vapour condensed in the intake manifold and in the sample line.

The EGR level was controlled by an EGR supply line with a gate valve (EGR valve). The EGR supply line (25.4 mm diameter) was insulated to reduce the heat loss of the exhaust gas. The EGR level was controlled by adjusting the EGR valve or the gate valve for the fresh air intake (air intake valve). The EGR level in the intake manifold was calculated by measuring the amount of CO_2 in the intake manifold and dividing it by the amount of CO_2 in the exhaust [see equation (1)].

For theory verification, the EGR was also measured using the temperature method described above. For this method, the intake temperature T_{air}, EGR temperature T_{EGR}, mixture temperature T_{mix}, and exhaust temperature T_{exhaust} were measured with thermocouples. All thermocouples used were type K and were wired to a thermocouple multi-channel digital temperature indicator (Omega DP462). The T_{mix}, T_{air}, and T_{EGR} thermocouples were probe-type thermocouples with exposed junctions (Alltemp A6-16-K-ESS), and the T_{exhaust} thermocouple had a probe with a grounded junction (Omega KQSS-18G).

During testing, the EGR temperature measurement was always lower than expected. This was due to intake air mixing into the EGR supply line. During the exhaust stroke of the single-cylinder CFR engine, the inlet and exhaust valves are both open for a brief time. While the inlet valve is open, the exhaust gas in the cylinder is pushed out of the inlet valve. This displaces the air in the intake manifold, causing the intake gas mixture near the EGR valve to be pushed into the EGR line (Fig. 6). For the model described above, the temperature measurements must be made outside the mixing region. Thus, for experimental purposes the intake temperature measurement was located outside of the mixing region. Also, the EGR temperature measurement (T_{EGR} in the figure) could not be used because of mixing; so the exhaust temperature measurement (T_{exhaust} in the figure) was used for the measurement of the EGR temperature. As discussed above, the model assumed an adiabatic mixing region. This assumption will not be valid since the mixing region is relatively large with a large surface area. The large surface area will result in high amounts of heat loss. This will cause the EGR measurements to be lower than expected since the

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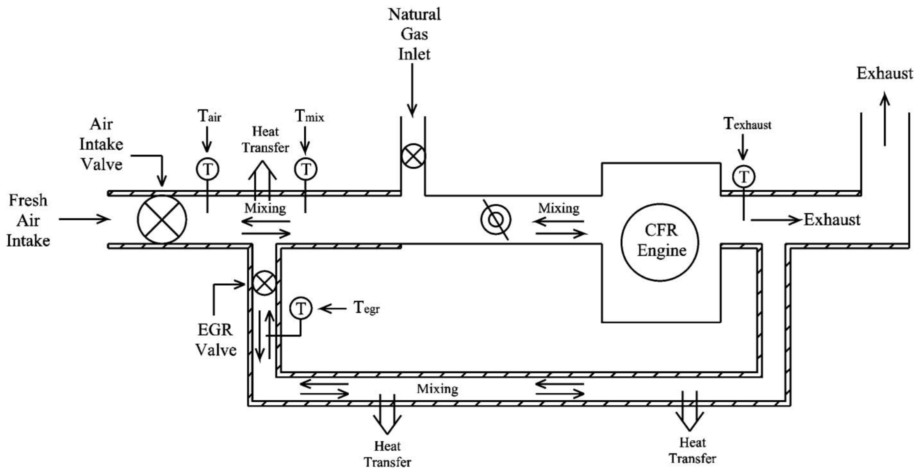

Fig. 6 Diagram of exhaust gas mixing and heat loss in the experimental set-up

temperature of the intake mixture will be lower owing to the heat transfer from the EGR line. Insulation was added to the EGR line and mixing region to reduce the heat transfer. The temperature measurements must be taken at steady state since changes in EGR levels will change the thermal equilibrium and heat transfer rates, resulting in inaccurate measurements in transient conditions. For a non-adiabatic model, equation (3) must be expressed as

$$\dot{m}_{\text{EGR}} h_{\text{EGR}} + \dot{m}_{\text{air}} h_{\text{air}} = \dot{m}_{\text{mix}} h_{\text{mix}} + \dot{Q}_{\text{out}} \tag{11}$$

where $\dot{Q}_{\text{out}}$ is the heat transfer rate from the mixing region into the environment. During transient EGR changes, the heat transfer rate $\dot{Q}_{\text{out}}$ will also change. The measured EGR level will differ from the actual EGR level if the heat transfer rate does not remain constant. Once the system reaches thermal equilibrium, the heat transfer rate will be constant, allowing for a stable EGR level measurement.

The DAWPD sensor was attached to the intake manifold with insulated tubing. The placement of the sampling point was located very close to the mixture temperature T_{mix} thermocouple. Ideally, the prototype sensor would be placed directly in the intake manifold; however, the prototype sensor was too large to be placed in the intake manifold. In a commercial application, a microelectromechanical system (MEMS) sensor could possibly be used to fit into the intake manifold. For example, capacitive micromachined ultrasonic transducers (CMUTs) can be manufactured with small size, high operating frequency, and wide temperature compatibility range and can replace the functions provided here by the larger slower piezoelectric transducers [**8**].

The gas mixture in the intake manifold was drawn through the prototype sensor by the pump in the gas analyser. The sensor output voltage and the temperature voltage were recorded for each EGR level. Since the intake mixture was drawn through the tubing and sensor, the temperature in the sensor would be lower than that measured by the T_{mix} thermocouple. This resulted in a less sensitive measurement because there was a smaller temperature change for a change in EGR level.

Resonant frequency changes of the ultrasonic transducers will cause phase shifts (and error) in the DAWPD sensor. The frequency ratio, the ratio of the driving frequency to the ultrasonic transducer's resonant frequency, determines the phase angle between the two signals [**9**]. The resonant frequencies of the ultrasonic transducers used in this experiment are temperature dependent. Therefore, changes in temperature will cause changes in the frequency ratio and in the phase angle. Since the DAWPD sensor measures the phase difference between the sent wave and the received wave, changes in phase angle will result in error. To compensate for the change in frequency ratio, the driving frequency was changed to match the resonant frequency for each measurement. The correlation between the resonant frequency and temperature was found by measuring the resonant frequency for various temperatures, where the resonant frequency was found by changing the driving frequency of the transmitting transducer to maximize the amplitude of the signal from the receiving transducer. From these measurements, a curve was found which described the resonant frequency as a function of temperature. This curve is shown in Fig. 7. For each measurement with the prototype sensor the temperature in the sensor was recorded; then the driving frequency was adjusted to match the resonant frequency of the transducers using the resonant frequency curve found in Fig. 7. After the frequencies were matched, the output

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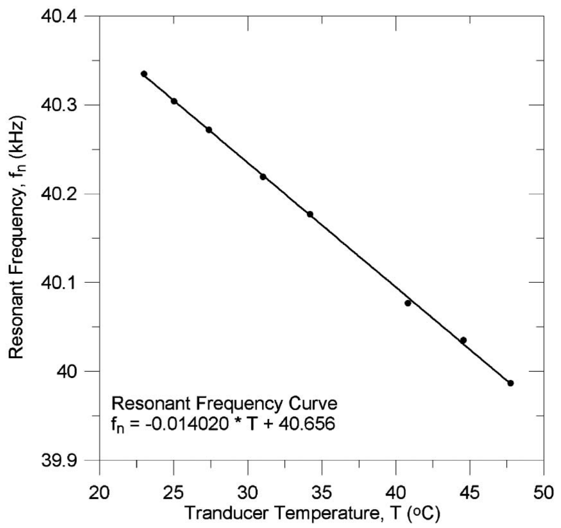

Fig. 7 Temperature dependence of the resonant frequency of the ultrasonic transducers

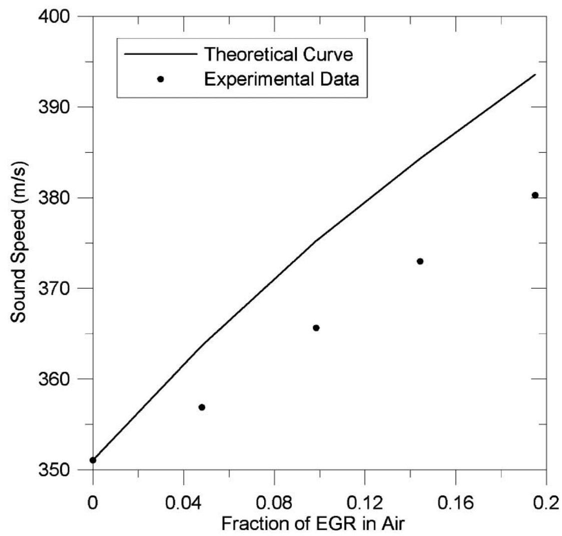

Fig. 8 Comparison of the theoretical response and the experimental data for the prototype EGR sensor

voltage of the sensor could be recorded. It should be noted that this complexity is due to the temperature sensitivity of the piezoelectric devices used in the prototype DAWPD sensor. CMUT devices with less temperature sensitivity would avoid most of the required adjustments [**8**].

4 EXPERIMENTAL RESULTS

The theory of EGR measurement was verified using experimental results. The experimental results show that the DAWPD method can be used to measure EGR with adequate accuracy and range.

A convenient method to compare the theoretical response and experimental measurements was to calculate the sound speed of the intake mixture as a function of the fraction of EGR. Figure 8 compares the experimental measurements and theoretical response. The theoretical sound speed is calculated by equation (10), where T_{exhaust} is used for T_{EGR}. The sensor output voltage can be related to the sound speed by calibrating the prototype sensor with gases of known sound speed (a mixture of nitrogen and methane for instance). As expected, the experimental data are consistently lower than the theoretical data. As discussed in the experimental set-up, the heat transfer from the exhaust line to the environment will result in a lower mixture temperature than is predicted by the adiabatic model. The sound speed of the intake mixture will also be lower because the mixture temperature is lower [see equation (9)].

The static characteristics of the prototype sensor that were examined were accuracy, repeatability error, calibration error, resolution error, and range. The property of interest for the EGR sensor is the concentration of EGR in the intake mixture. Therefore, the sensor characteristics will be expressed in percentage of EGR in the exhaust by mass. These static characteristics are summarized in Table 2. The calibration curve of the prototype EGR sensor is shown in Fig. 9. For the calculation of the calibration, repeatability, and resolution error, the most conservative, or worst-case, error limit was used. The maximum error measured for each characteristic is given and the estimation of the accuracy of the sensor is the r.m.s. value of the calibration, repeatability, and resolution error. The accuracy of the EGR sensor was found to be ± 1.3 per cent EGR or a relative uncertainty of ± 6.5 per cent full scale. It is unclear what accuracy would be necessary for optimum EGR control. However, it is expected that this accuracy would be adequate for this application.

The range of the prototype was limited by the maximum operating temperature of the piezoelectric transducers used in the prototype. The transducers can only be operated to a maximum temperature of 60 °C. At high EGR levels the temperature of the intake mixture can be much higher than the maximum

Table 2 Static characteristics of the EGR sensor

Characteristic	Value (per cent EGR)
Accuracy δ_{total}	± 1.3
Calibration error δ_{cal}	± 0.62
Repeatability error δ_{rep}	± 0.64
Resolution error δ_{res}	± 0.95
Range (as tested)	0–20

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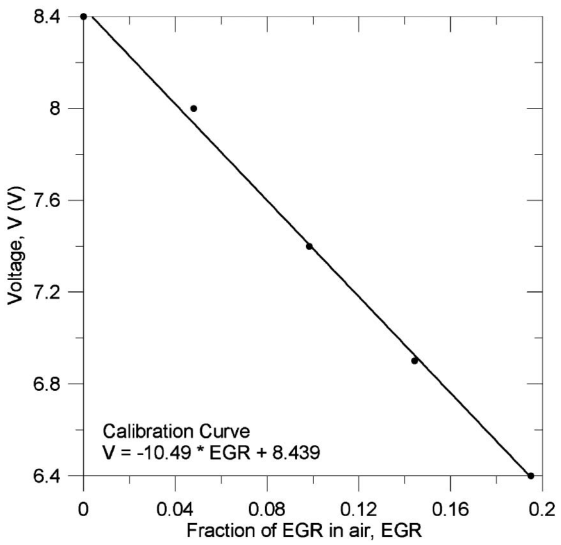

Fig. 9 Calibration curve of the prototype EGR sensor

operating temperature. At EGR levels of 20 per cent the temperature in the sensor reached 52 °C, which is relatively low considering that the temperature in the intake manifold was 97 °C. Therefore, the experimental testing was limited to 0–20 per cent EGR levels. For commercial applications the sensor would be placed directly inside the intake manifold, where the temperature can reach 250 °C with high levels of EGR. Therefore, different ultrasonic transducers that could operate over a range of temperatures from −40 to 250 °C would be required.

5 IMPROVEMENTS FOR A COMMERCIALLY SUCCESSFUL EGR SENSOR

Further modifications are needed to make a commercially successful EGR sensor. The current EGR sensor prototype was limited by its size, temperature rating, temperature dependence, and relatively large air and EGR mixing section. The sensor performance could possibly be improved by using microsized high-temperature MEMS ultrasonic transducers in a near-adiabatic mixing section. CMUT devices exhibit a relatively wide temperature range and low temperature sensitivity, making them more ideal for this purpose than the currently available piezoelectric transducers [**8**].

Firstly, the size of the prototype sensor did not allow it to be placed in the intake manifold. MEMS transducers may be required to make a sensor small enough to fit inside the intake manifold. Placing the sensor inside the manifold will result in a more accurate sensor since the sensitivity of the sensor will increase because the intake mixture temperature will be higher. Secondly, the natural frequencies of the transducers that were used in the prototype were temperature dependent. If not compensated for (by using resonant frequency matching, as was done here), this can result in large errors as the natural frequency deviates from the driving frequency. For a commercial sensor, MEMS transducers could be used so that the transducers are not driven near resonance, so that changes in frequency ratio will only produce small phase angle changes [**7**]. Thirdly, the range of the prototype sensor was limited by the maximum operating temperature of the ultrasonic transducers. MEMS transducers can have high operating temperatures, allowing the prototype sensor to be operated in the high-temperature conditions that exist in an intake manifold. Fourthly, the intake air and EGR gas mixing section should be small enough that it could be considered adiabatic. In the experiment, back flow from a single-cylinder CFR engine caused mixing throughout the EGR line. In multi-cylinder engines the back flow will be considerably less because the back flow from one cylinder will be consumed by another cylinder starting the intake stroke. Therefore, the mixing section will be small allowing measurements to be made in real time instead of requiring a steady state thermal equilibrium measurement for adequate accuracy.

6 SUMMARY

EGR sensors can be used to improve the performance of internal combustion engines. Various EGR measurement methods can be used; however, a durable, fast-response, and cost-effective method is the DAWPD acoustic technique. The DAWPD sensor was able to measure EGR in steady state conditions with adequate accuracy (±1.3 per cent EGR).

Modifications must be made for a commercial sensor to be developed. These modifications possibly include the following:

(a) MEMS transducers for a smaller sensor to fit inside the intake manifold;
(b) transducers with little resonant frequency change or possibly non-resonant devices;
(c) transducers which can operate in high-temperature environments;
(d) operation in a small mixing section so that measurement could be made in real time.

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APPENDIX

Notation

c	speed of sound
C_p	specific heat at constant pressure
C_v	specific heat at constant volume
h	specific enthalpy
$\dot{m}$	mass flowrate
M	molecular weight
$\dot{Q}_{\text{out}}$	heat transfer rate
R	universal gas constant
T	temperature
w	mass fraction
x	mass ratio
y	volume fraction
γ	ratio of specific heat $= C_p/C_v$
Φ	equivalence ratio

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